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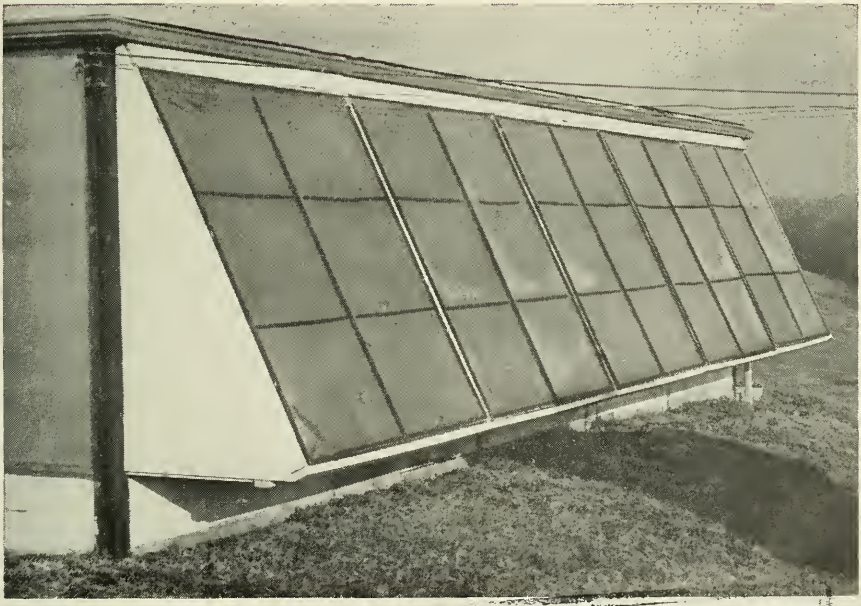
The Sun—A Poultry House Furnace



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Bulletin 548T June 1967
West Virginia University
Agricultural Experiment Station

The Sun—
A Poultry House Furnace

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SUMMARY

In 1961 the first solar collectors were built at West Virginia University by the Department of Agricultural Engineering. Three units were tested. A flat, a corrugated, and a curved surface gave efficiencies of 54.9 per cent, 52.1 per cent, and 52.9 per cent, respectively. All units had an area of 32.6 square feet and the energy transporting media was air.

In 1964 a 320 square foot, flat plate solar collector was placed on the south side of a pole panel poultry house. The poultry house environment was monitored and from the results the heat losses and gains were calculated.

Finally, in 1965 other collectors were designed. A double-plated, a finned collector, and flat-plate collector were tested. These newer collectors have not been used on a building.

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The Sun— A Poultry House Furnace

Oscar R. Harman

THE world's main source of energy is the sun. Scientists estimate the total sunlight falling on the earth for about three days equals all the energy in the world's reserves of coal, oil, and wood, plus all atomic fuels.

One use of the sun's energy is supplemental heat for farm structures. Today it is recognized that the environment plays an important part in animal production. By using inexpensive solar collectors environment can more easily be controlled.

Longhouse (10) in 1960 built three different types of solar collectors. These collectors were made of aluminum and included a flat, corrugated, and a curved surface, painted flat black (Figure 1). Plastic was used to cover all the collectors.

Since winter is the time of year supplemental heat is needed, the collector plates were placed 64 dgrees from the horizontal. At this inclination the sun's rays strike the plate perpendicularly at solar noon in the winter in West Virginia. The back of the curved collector was sloped 80 degrees and the lower portion of the plate sloped 30 degrees from the horizontal.

In order to eliminate heat exchangers from the collectors, air was used as the transporting media for the heat. Centrifugal fans were used to drive outside air through a 10½-inch by 11-inch duct. The air, after passing through the duct, moved down under the collector plate and finally out of the solar collector.

Heat losses were reduced by use of mineral wool batts 3 inches thick placed in the walls of the collector structure. Also the air duct was insulated. All joints were caulked and the structure was painted white.

In order to evaluate the collectors, Longhouse (10) used a 16-point Brown recording potentiometer along with copper-constant thermocouples to measure the outside and exhaust temperatures of the air. An AC-DC hygrometer converter connected an Aminco-Dunmore relative humidity (RH) sensing element to the Brown recorder. In this manner the RH of the outside air was measured.

Before the fans were placed in the collector structure, they were placed on a wind tunnel and a static pressure vs. cfm performance

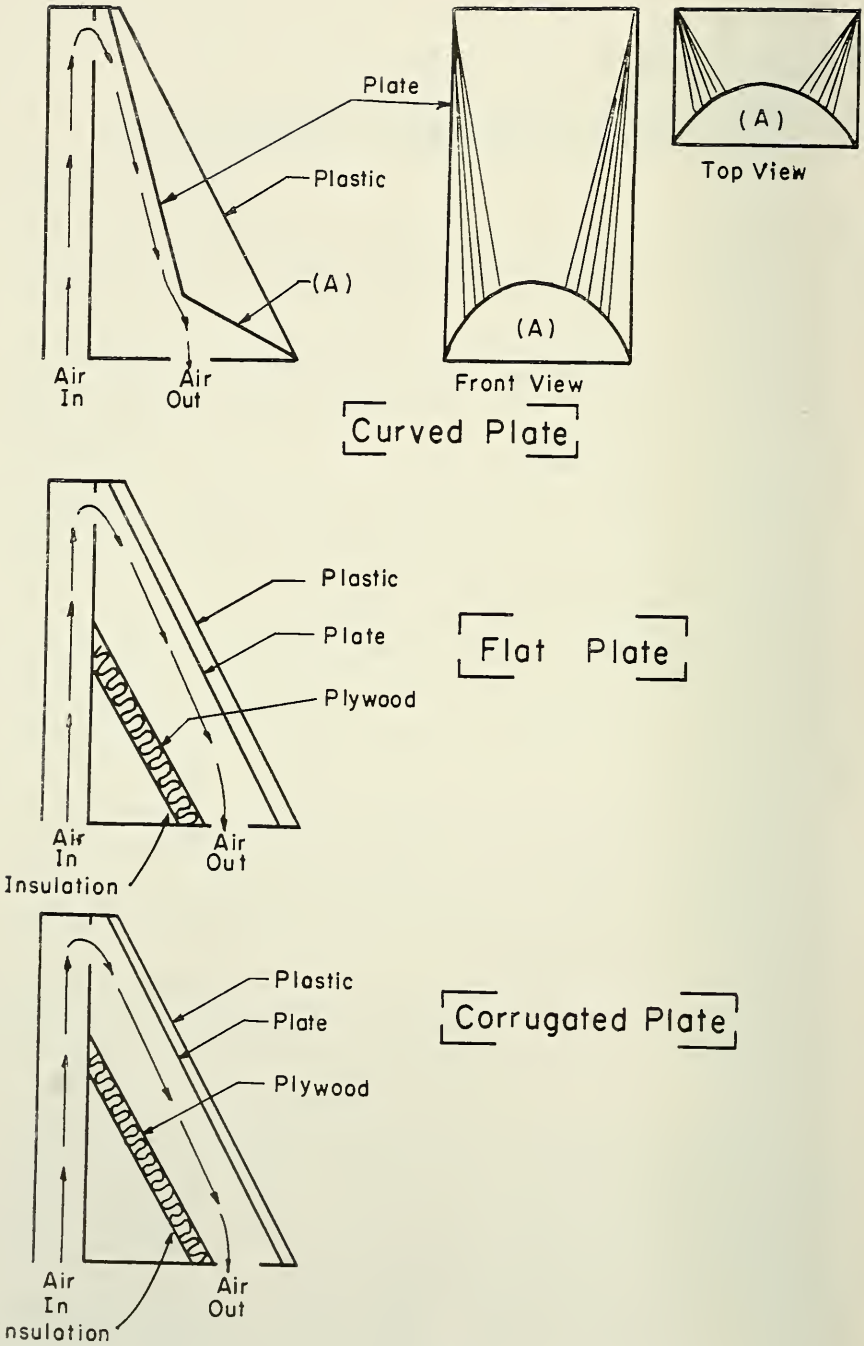

Figure 1. Cross sections of solar collectors used by Longhouse (10).

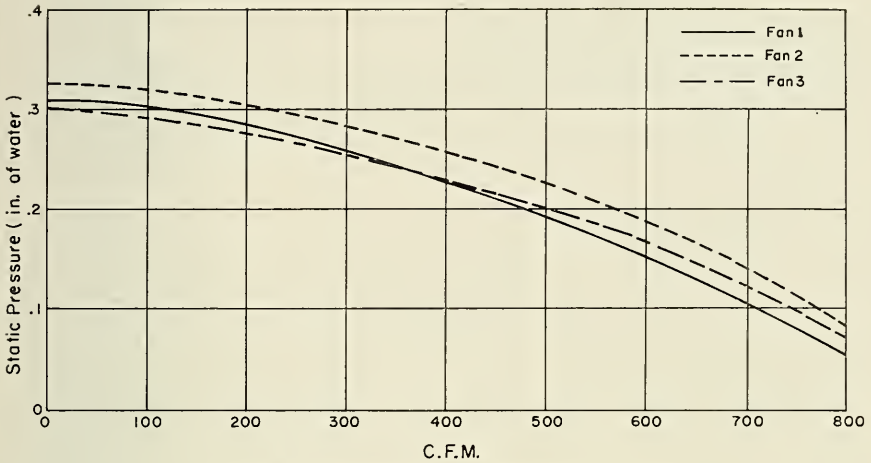

Figure 2. Fan performance curves (10).

curve was developed (Figure 2). The procedure followed was in accordance with the National Fan Testing Code. The air flow could then be determined by measuring the static pressure in the air duct.

A Beckman and Whitney thermal radiometer was used to measure the solar radiation. This unit was placed by the collectors and at the same inclination as the collector plate.

All calculations were made on an hourly basis. The insolation rate was calculated by the following formula (10):

$$Ir = Mv\ (C)\ (K) \qquad \text{Equation 1}$$

Ir = insolation rate in Btu/hr-square feet
Mv = millivolts recorded per hour
C = calibration correction factor for radiometer plate temperature
K = constant for radiometer of 24.82 Btu/hr-square feet-mv.

Using the above information the insolation rate could be calculated and was plotted.

The energy collected was calculated by the following equation (10):

$$Q = \frac{cfm\ (Cp)\ (\Delta t)\ (M)}{WA} \qquad \text{Equation 2}$$

Q = solar heat collected, Btu/hr square feet
cfm = air flow
Cp = specific heat of air = 0.24 Btu per pound °F
Δt = temperature difference between exhaust and incoming air
M = minutes fan operated per hour
W = cubic feet per pound of dry air
A = effective area of transmittance.

With the solar insolation, the energy collected by the plates was calculated and plotted (Figure 3).

Some heat energy was lost through the walls of the collector structure. This loss was calculated by the equation:

$$q = UA_s\ (\Delta t) \qquad \text{Equation 3}$$

where

q = heat rate, Btu/hr

A_s = area, ft²

Δt = temperature difference of outside and inside air, °F

U = overall thermal transmittance, Btu/hr square feet °F

The value of U was .132 Btu/hr ft² °F and the area was 15.1 square feet. The heat loss was calculated and added to the collectors' rate.

The data for the tests are shown in Figures 3, 4, 5, and 6. The first test was June 17, 1961, and shows the insolation rate to dip between 6 a.m. and 8 a.m. Fog moved over the area of the collectors and caused a decrease in energy at the earth's surface. The insolation rate peaked at noon with a value of 241.7 Btu per square foot per hour.

The corrugated plate collected the most energy at noon; however, for the day, it absorbed 894.0 Btu/day-ft². The curved plate collected

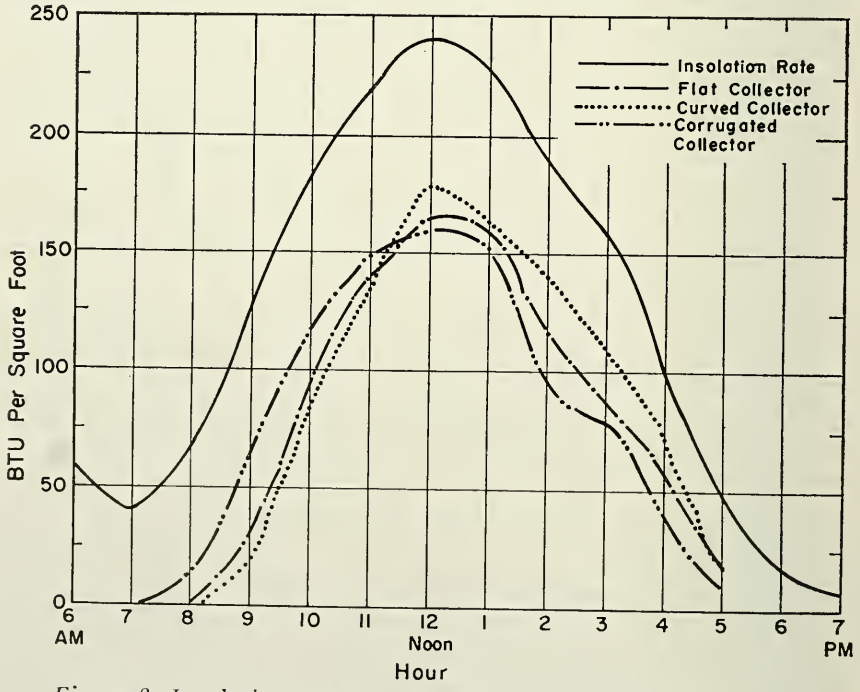

Figure 3. Insolation rate and heat gains through collectors on June 17, 1961 (10).

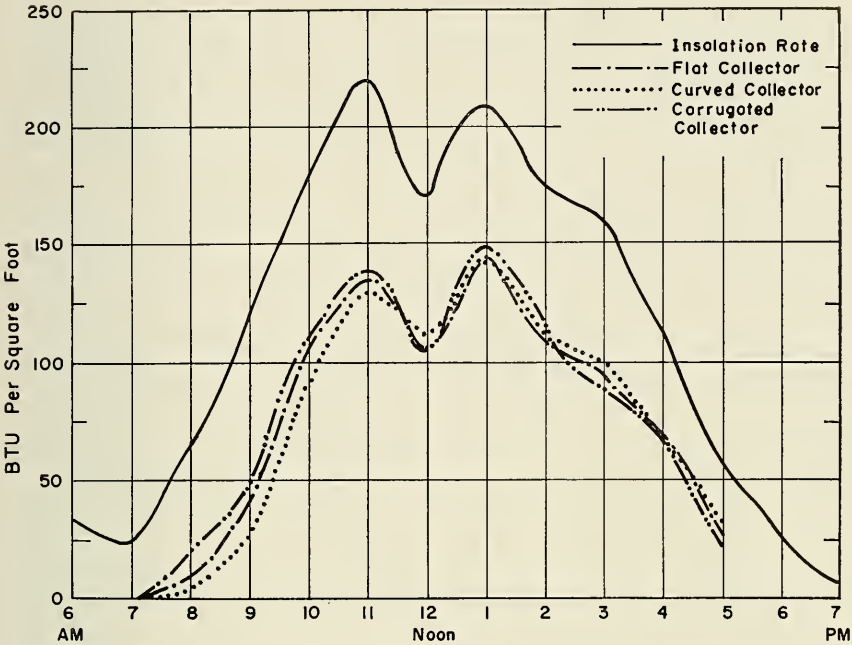

Figure 4. Insolation rate and heat gains through collectors on June 18, 1961 (10).

928.2 Btu/day-ft^2. The efficiencies for the collectors are shown in Table 1.

TABLE 1

Efficiency of collector on June 17, 1961

Collector	Efficiency (Per Cent)
Curved	54.9
Corrugated	52.9
Flat	52.1

All collector plates for these tests had an area of 32.6 square feet. The air flow varied through the collectors from 465 cfm for the curved unit to 645 cfm for the flat unit and 590 cfm for the corrugated unit.

The second test on June 18, 1961, was again affected by fog at 7 a.m. (Figure 4). Between 11 a.m. and 1 p.m. the sky became cloudy and a dip was evident on the insolation rate curve as well as on all the collector curves. This is an example of the effect of clouds on the energy collected. Table 2 shows the characteristics of the collectors.

A third test was made on June 22, 1961, (Figure 5) to study the effects of clouds on the amount of energy collected. The collectors

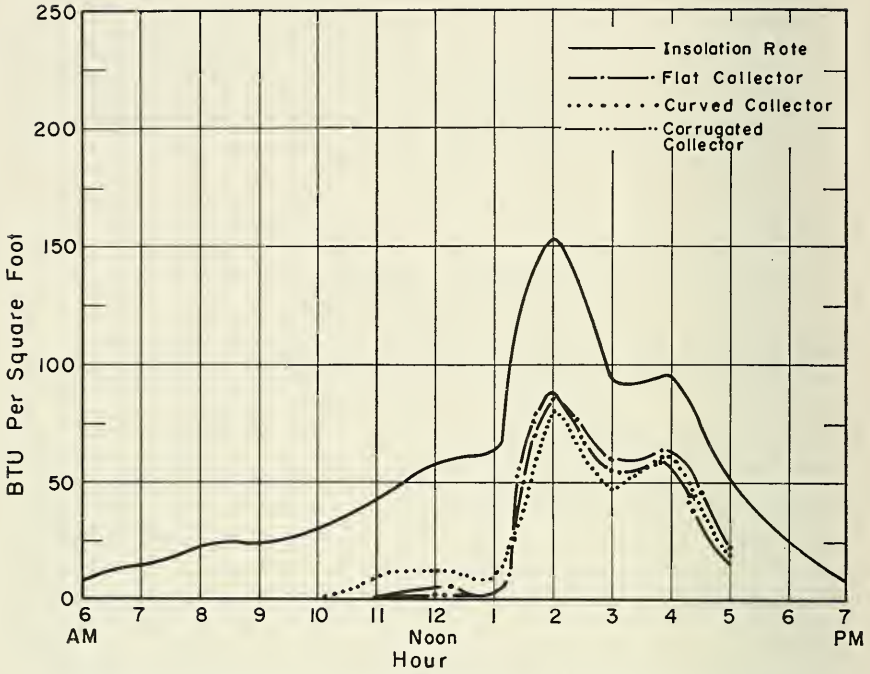

Figure 5. Insolation rate and heat gains through collectors on June 22, 1961 (10).

show a close similarity in this test with total radiation of 694.9 Btu per square foot, and energy collected of 244.4 Btu, 239.0 Btu, and 225.4 Btu per square foot for the curved, flat, and corrugated collectors. Efficiencies were 35.1, 34.3, and 32.4 per cent for the curved, flat, and corrugated collectors respectively (10). On cloudy days more radiation reaches the earth in a diffused state than on sunny days; therefore, the geometry of the receiver would not be as important on cloudy days.

The final test was made on June 27, 1961, (Figure 6). The day was clear except at 8 a.m. and 2 p.m. At 8 a.m. a drop occurred in the insolation curve but none appeared in the curves of the collectors.

TABLE 2

Performance of collector on June 18, 1961

Collector	Efficiency (Per Cent)	Radiation Absorbed (Btu/day-sq ft)	Area ft^2
Curved	52.7	823.7	32.6
Flat	53.8	840.1	32.6
Corrugated	55.3	863.7	32.6

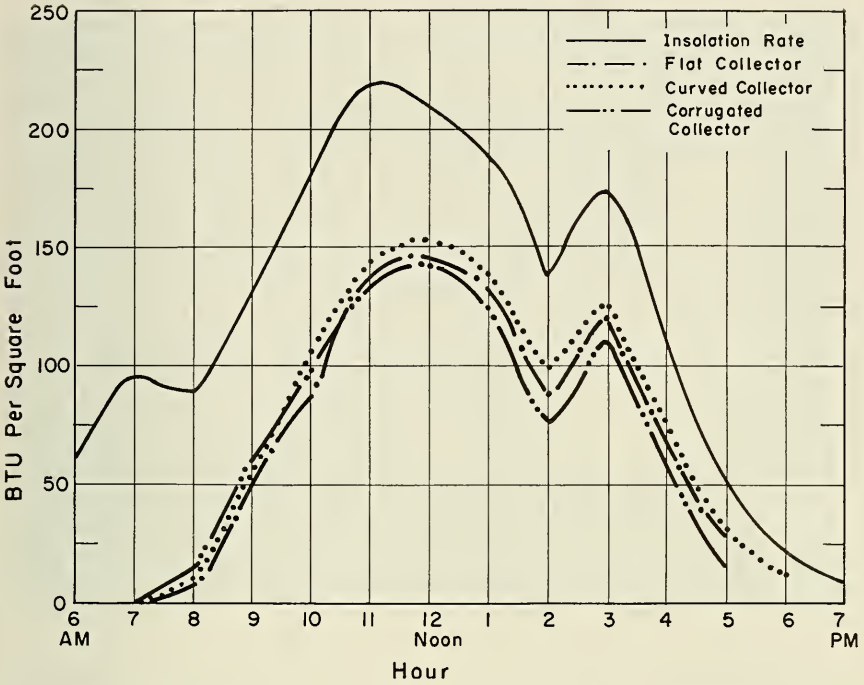

Figure 6. *Insolation rate and heat gains through collectors on June* 27, 1961 (10).

This effect is probably due to the collector structure giving up heat energy to the air and no dip was measured. The 2 p.m. dip was due to reduced radiation received at the collectors. The analysis is shown in Table 3.

TABLE 3

Solar collector performance June 21, 1961 (10)

Collector	Radiation Collected (Btu/ft² day)	Efficiency (Per Cent)	Total Radiation (Btu/ft² day)	Air Flow (cfm)
Curved	949.0	56.4	1682.9	465
Flat	879.0	52.2	1682.9	465
Corrugated	835.5	49.6	1682.9	590

COLLECTORS ON A POULTRY HOUSE

In order to determine the effects of solar collectors on the environment of a poultry house, a flat plate collector was placed on the south side of a 32′ x 40′ building (Figure 7). The building was a pole panel type structure and was insulated with 3″ mineral wool in the sides and 6″ in the roof section (6).

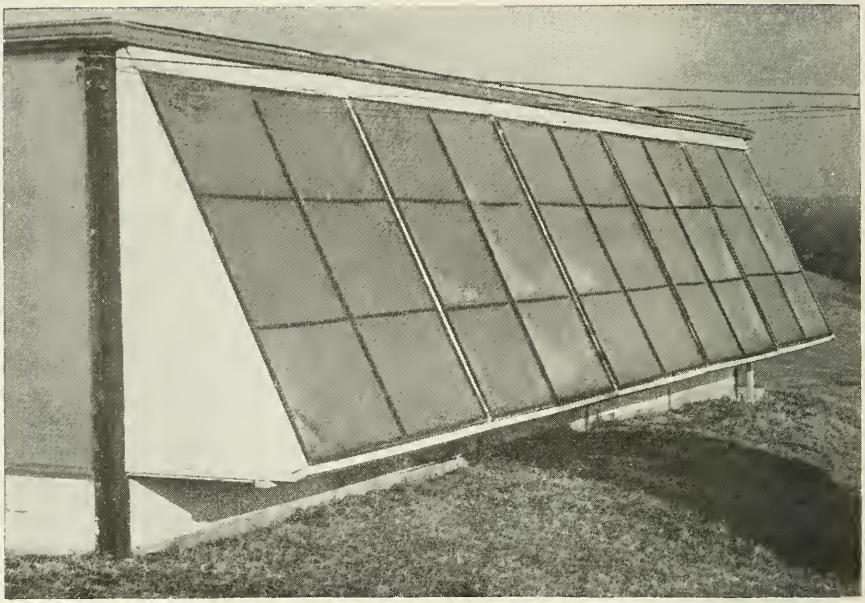

Figure 7. Solar collectors installed on the poultry house.

Ventilation was provided by three fans located under the solar collectors. The fans were thermostatically controlled; one thermostat was placed under the collector plate and the other inside the building. The controller under the plate turned the fans on if the air temperature was 65° or higher, thus moving the energy collected into the building in the winter. If the temperature inside the building was 65° or above, the inside controller would operate the fans cooling the building.

As stated earlier, a flat plate solar collector was placed on the south side of the building. Five panels 8′ x 8′ gave a total of 320 square feet of collecting area. The plate of the collector was aluminum with the top painted black. The under side was painted with aluminum colored paint. Covering the collector plate was a plastic sheet approximately 1″ from the plate.

The wall of the poultry house, the collector plate, and solar collector doors formed a triangular air space along the length of the building (Figure 8). In order for air to move up under the solar collector, a 6″ wide opening was made to the outside. This opening ran the full length of the house just behind the collector plate. When the fans were operating, the air moved from the outside through the 6″ opening, up along the back of the collector plate and into the building.

The major heat gains are produced by the birds and the solar collectors. In this analysis the average of 15 readings per hour was taken as the temperature inside the house. Knowing this temperature, the

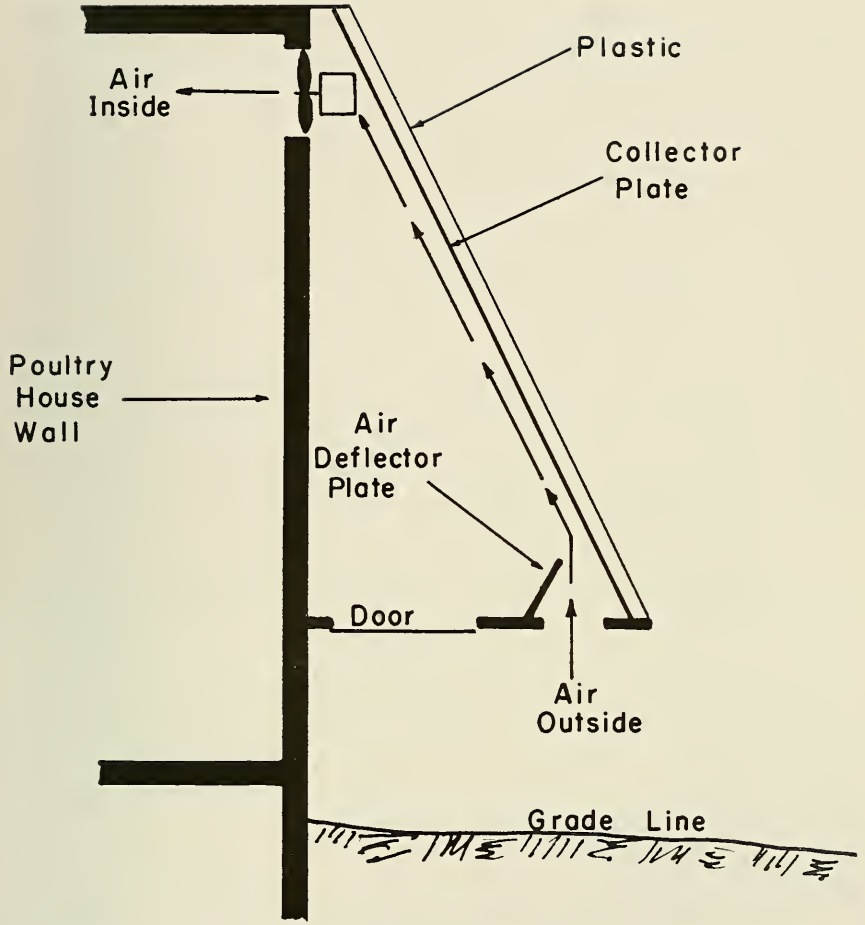

Figure 8. *Cross section of solar collector on the poultry house.*

Btu/lb of bird was found from Ota's data (12). Multiplying the average bird weight by Btu/lb gave the heat produced per bird. Finally, the total heat produced by the birds could be calculated by multiplying the Btu/bird by the number of birds.

The average heat production of the bird was used in this analysis. The bird will produce less heat during the night than during the day; however, the bird is able to maintain its body temperature by the position of the feathers.

An example of how the heat produced by the bird was calculated follows:

Temperature inside house = 55° F
Number of birds = 544
Bird weight (average) = 4 lb

From Ota's data (12) it is seen that at 55° F the bird produces 9 Btu/lb. A 4-pound bird would produce:

$$9 \times 4 = 36 \text{ Btu/bird.}$$

544 birds were housed in the house under study.

Therefore:

$$36 \times 544 = 19{,}584 \text{ Btu/hr}$$

and

$$19{,}584 \times .75 = 16{,}788 \text{ Btu/hr sensible heat.}$$

The heat produced by the birds in the house equals 19,584 Btu/hr, for an inside temperature of 55° F.

Some heat is produced by the lights in the building. The energy used by the lights was measured by a kilowatt hour meter and found to be 7.4 kilowatt hours per day, or .53 kilowatt hours per hour for 14 hours. In order to covert the .53 to Btu per hour, it must be multiplied by 3,415 Btu per kilowatt hour. Thus the heat produced by the lights was calculated by multiplying:

$$.53 \times 3{,}415 = 1{,}800 \text{ Btu/hr.}$$

The fans in the ventilation system provided another source of heat. The energy used by the fans was metered by kilowatt hour meters. Tests were conducted on the motor-fan system to determine the efficiency and air flow, and the efficiency of the system was found to be 14 per cent. This means that 86 per cent (100-14) of the input energy was put into the air stream as heat. To calculate the heat input by the fans, the kilowatt hr/hr must be multiplied by .86 to determine the amount of energy converted to heat. The following formula was used to calculate the heat input due to the fans:

$$Q = 3{,}415 \text{ K B} \qquad \text{Equation 4}$$

Where

Q = Btu/hr, heat produced by fans

K = kilowatt hrs/hr, energy used by fans

E = over all efficiency of fan system = .14

and

$$B = 1 - E.$$

The solar collector was the other heat source for the building. Using equation 2 the heat produced by the solar collector can be calculated. Δt in the equation represents the temperature difference of the outside air and the air under the solar collector. In using this equation, it is important that the true air temperature under the solar collector is measured. Therefore, an average temperature from 15 readings per hour is used in the calculation. Also 8 thermocouples connected in paralled acted as an averaging sensing device.

The major heat loss is through ventilation. Air was pushed through the house by 3 fans, each having a capacity of 750 cfm. This ventilation

heat loss can be calculated from the following equation:

$$Q = \frac{(V)\ (Cp)\ (\Delta t)}{W} \qquad \text{Equation 5}$$

Where

Q = Btu/hr lost through ventilation
V = volume of air moved per hr (ft^3)
Cp = specific heat = .24 Btu/ (pound °F)
W = cubic feet per pound of dry air
t = temperature difference of incoming air and outgoing air (°F).

The volume of air moved is known by knowing the capacity of the fans and their running times. The value of W must be determined. In this analysis W was determined from the psychrometric chart for the conditions measured. Temperature inside and temperature outside were measured by thermocouples and a Brown potentiometer.

The heat loss due to conduction can be calculated by the equation (4, 5, 11):

$$Q = UA\ (t_i - t_o) = UA\Delta t \qquad \text{Equation 6}$$

Where

U = over all coefficient of heat transfer
Q = heat lost by conduction
A = wall or roof area
t_i = temperature inside
t_o = temperature outside
$\Delta t = t_i - t_o$

and

$$U = 1/R \qquad \text{Equation 7}$$

Where

$R = 1/f_1 + X_1/K_1 + X_2/K_2 + X_3/K_3 + X_4/K_4 + 1/f_2$
R = resistance to heat flow, total across the section
f_1 = film conductance, inside
f_2 = film conductance, outside
$X_{1, 2, 3, 4}$ = thickness of material in wall or roof section
$K_{1, 2, 3, 4}$ = thermal conductivity (Btu in per hr sq ft °F).

The wall section consists of the following:

.025 inch embossed aluminum
3 inch mineral wool
¼ inch exterior plywood.

The roof section consists of the following:

½ inch celotex
6 inch mineral wool
¼ inch exterior plywood
2 layers 55 lb roofing.

The value of R, using film coefficients for the wall sections of 1.65 inside and 6.00 outside, gave the following:

$R = 1/1.65 + .025/119 + 3/.27 + .31/1 + 1/6.00$

$R = 12.182$

$U = 1/R = .083$ Btu/ (hr sq ft °F).

Using the same film coefficients for the roof as were used for the wall, the roof U value was calculated as:

$R = 1/1.65 + .5/0.34 + 6/.27 + 2/6.5 + .31/1 + 1/6.00$

$R = 25.081$

$U = 1/R = .0398$ Btu/ (hr sq ft °F).

Values of K were from Jennings and Lewis (5) and also the Heating, Ventilating, Air Conditioning Guide (1959).

A U value of .24 Btu/ (hr sq ft °F) was listed (11) for 8" x 8" x 16" insulation filled concrete block. Again the film coefficients must be added to give a U value of .22 Btu/ (hr sq ft °F) for the concrete block.

List of U values

Section	U
Roof	.04
Wall	.08
Concrete Block	.22

A feed room was added to one end of the poultry house. This means that not all the wall area is exposed to outside temperatures. Jennings and Lewis (5) recommend the temperature difference for such conditions be calculated from the following formula:

$\Delta t = (t_i - t_o) \times 0.5.$

Also the air velocity inside the feed room was considered as still air. The insulation value of still air is greater than moving air so the U value of this wall section should be .079 instead of .082 Btu/ (hr sq ft °F). The concrete block between the feed room and the poultry house will have a different U value also. The U value of the block was calculated as follows:

$R = 1/1.65 + 1/.24 + 1/1.65$

$R = 4.92$

$U = 1/R = .20$ Btu/ (hr ft² °F).

The doors (two) between the feed room and poultry house consist of two ¼" plywood sheets on a 2" x 4" member frame. The U value for the doors was found to be .35 Btu/ (hr ft² °F).

The areas, A, were calculated as follows:

Door area = 3' x 8' x 2 = 48 ft²

Block = 1½' x 32' − 2 (1½' x 4') = 48 − 12 = 36 ft²

Other = 8' x 32' − 2 (4' x 6½') = 256 − 52 = 204 ft²

Wall = (8′ x 32′) 3 = 768 ft²
Roof = 32′ x 32′ = 1024 ft²
Concrete Block = 1½′ x 32′ x 3′ = 144 ft².

The temperatures, t_i and t_o, were measured with thermocouples as the sensing elements and a Brown Potentiometer as the recorder. (See Appendix for data.)

The collectors on the south side of the building were given no consideration as insulation. The collector protects the wall from the wind, but at times the fans are operating and drawing cold air across the wall; therefore, it is assumed the solar collectors have no insulation value.

A considerable heat loss takes place by infiltration and air leakage through cracks around the doors and gutter cleaner openings. Only an estimate can be made as the amount of loss is indeterminate. Neubauer and Walker (11) recommend 1 to 1½ air changes per hour. The 1959 Heating, Ventilating, Air Conditioning Guide (4) gives a value of two air changes per hour for a room exposed on three sides.

In this analysis an air change rate of 1¼ times per hour is assumed. Using the equation presented by Neubauer and Walker (11) the heat loss was calculated:

$$H = 0.018\ VN\ (t_i - t_o) \qquad \text{Equation 8}$$

Where

H = heat loss due to infiltration, Btu hr
V = volume of house, ft³
N = number of air changes per hour
t_i = temperature inside
t_o = temperature outside.

The volume of the house is 32′ x 32′ x 9½′ or 9,750 cubic feet. Therefore, the above equation can be stated as follows:

$$H = 219\ \Delta t = 219\ (t_i - t_o)\ .$$

Jennings and Lewis (5) state: "It is customary to consider the temperature of the ground water as indicative of the subsurface earth temperature." Since the ground water temperature in West Virginia is approximately 55° F and the inside temperature of the house was approximately 55° F, the temperature difference would be zero. Therefore, no heat loss or gain was considered through the floor.

Another possible heat source in the building was the litter. It has been reported (3), however, that litter below 45 per cent moisture (dry basis) generated negligible or no heat. Therefore, no heating value was assigned the litter in this analysis.

Figures 9 through 15 show the effects of the solar collectors on the poultry environment. Also an environment in a house without solar

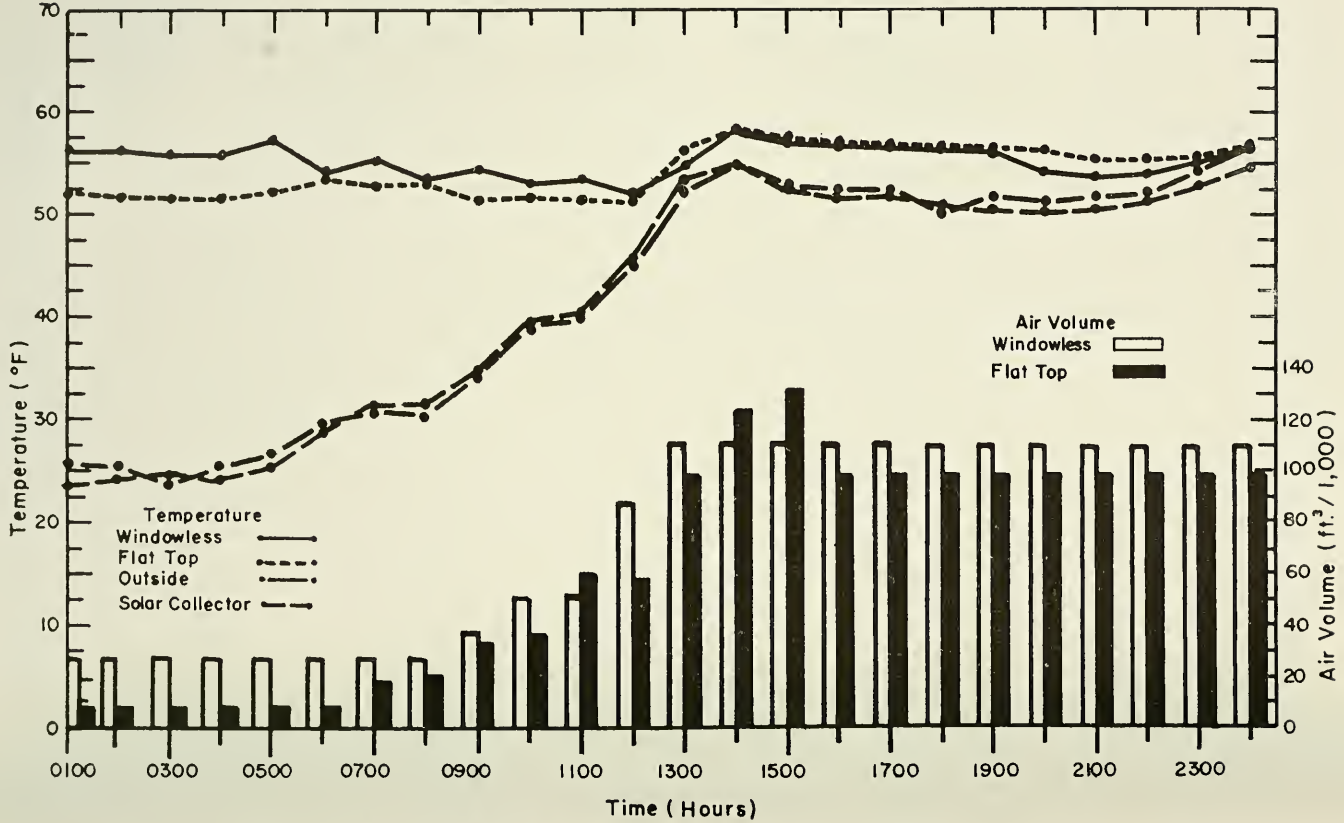

Figure 9. Effect of solar heat for a cloudy day on temperature and air flow (4-2-64).

collectors is plotted in Figures 9 and 12 to compare the thermal lag of the buildings. The house designated the "Flat Top" is the house with solar collectors. The windowless house is a poultry house with no solar collectors.

The first test was conducted on a cloudy day, April 2, 1964. There was little difference observed between the two houses (Figure 9). The air flow varied some but this was due to the thermostat setting, not the collectors. The air flow in both buildings was controlled by the outside temperatures and little effect is noticed by the use of solar collectors.

Figure 10 shows the heat losses from the poultry house with solar collectors on April 2, 1964. Infiltration and conduction losses were high in the morning, but as the outside temperature increased, these two losses decreased to zero by midnight. The ventilation losses increased at mid-day and showed the characteristics of on-off control.

Relative humidity on a cloudy day is compared to the inside conditions in Figure 11. The outside relative humidity is higher than the inside in the morning. However, the relative humidity inside the house becomes higher around noon and remains higher the rest of the day.

On April 4, 1964, tests were conducted upon both poultry houses. This day was a clear sunny day. In Figure 9 it was noted that the air flow increased in both houses to a maximum at the same time of day, but on a sunny day the air flow increased to a maximum sooner in the house with solar collectors (Figure 12). The time difference amounted to 5 hours for this day and means that more air is passed through the

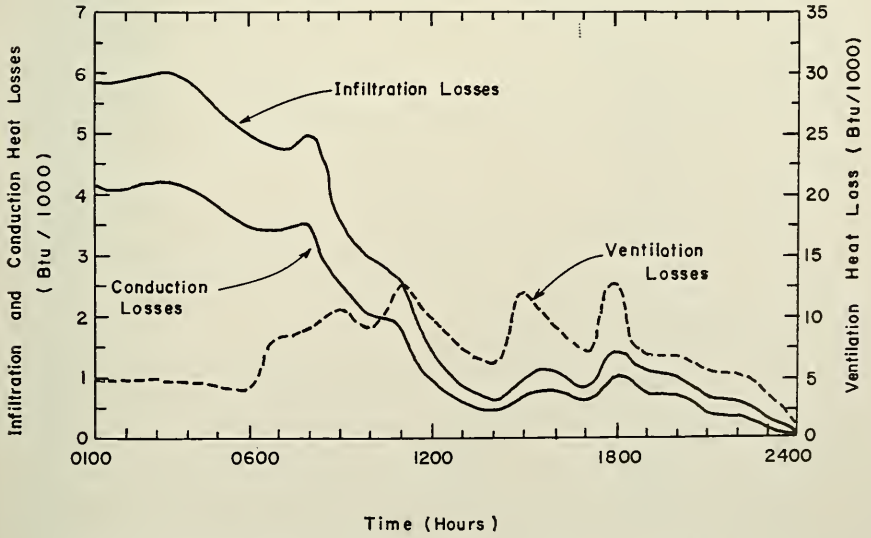

Figure 10. *Heat losses from poultry house on a cloudy day with an outside temperature rise* (4-2-64).

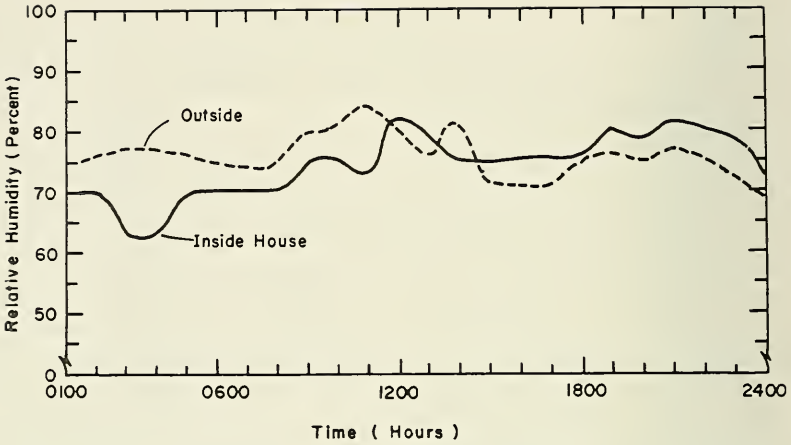

Figure 11. *Comparison of inside and outside relative humidity on a cloudy day* (4-2-64).

building with the collectors. From 7 a.m. to 5 p.m. the collector raised the temperature of the outside air, and from 12 a.m. to 5 p.m. a maximum air flow rate was observed.

The increase in air flow has many benefits. The air will remove ammonia from the building, moisture from respiration, and moisture in the litter.

Figure 13 relates the heat gained by the solar collectors to that lost by ventilation. In the morning the two curves show the same shape; however, in the afternoon the collector drops off evenly while the ventilation curve drops off in steps until evening and then it cycles. This behavior in the afternoon is due to the effects of the building acting as a heat sink; as the radiation from the sun decreases, the characteristics of the building override the effects of the collectors upon the building.

Comparison of inside and outside relative humidities are presented in Figure 15. The tables in the Appendix show measured relative humidity as well as the calculated data used in the figures.

OTHER COLLECTORS TESTED

After the flat plate collector was tested, different designs for solar collectors were tested to find a more efficient collector. Such factors as length of air passage, depth of passage, air flow rate, and type of collector surface were used to design more efficient collectors.

Two such collectors, along with the flat type collector, were tested in 1965 (Figure 16). The first collector was a twin corrugated plate type. The top plate had holes through which radiation could pass, as

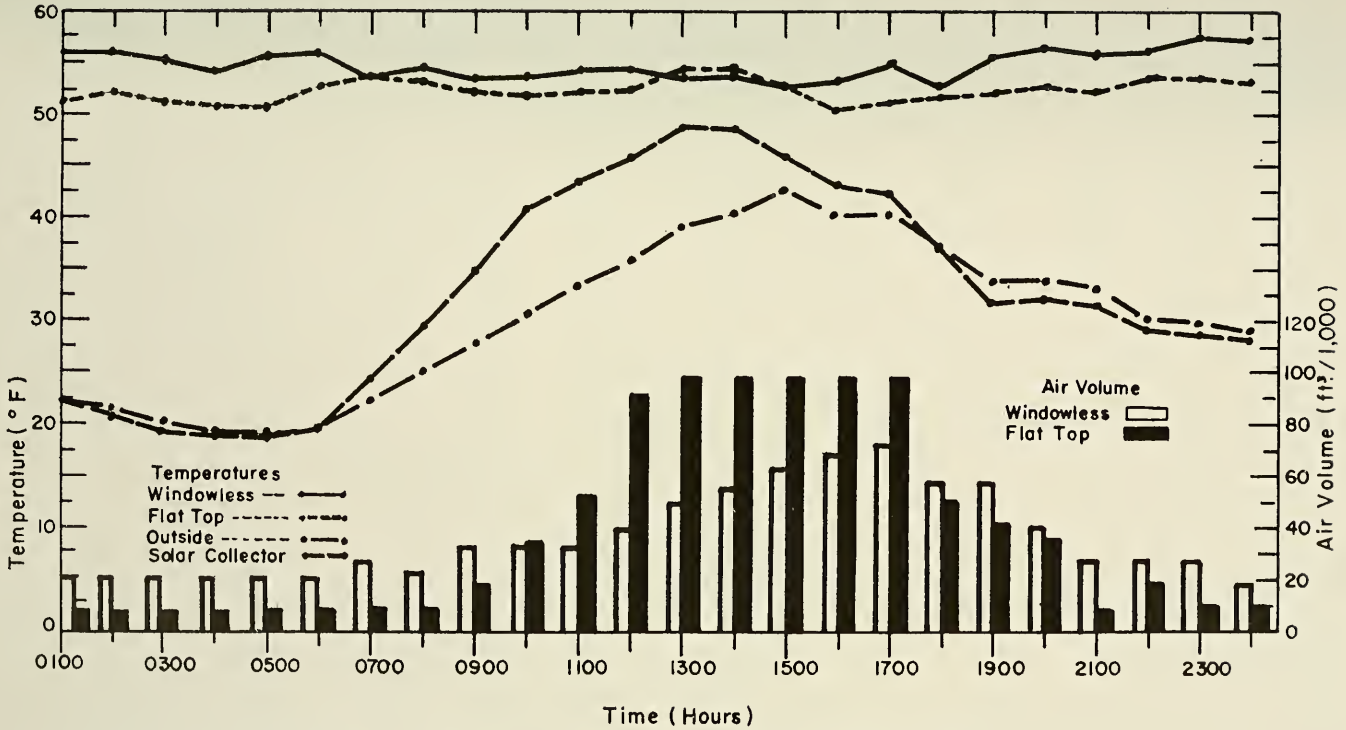

Figure 12. *Effect of solar heat for a clear day on temperature and air flow* (4-4-64).

well as to collect radiation from the top plate. The 1-inch diameter holes were punched in the corrugated aluminum approximately 3 inches apart on a square design. Both plates were corrugated and the valley portion of the bottom plate was placed under the valley portion of the top plate. This design permitted a longer path for the air to move through the collector thus increasing the temperature.

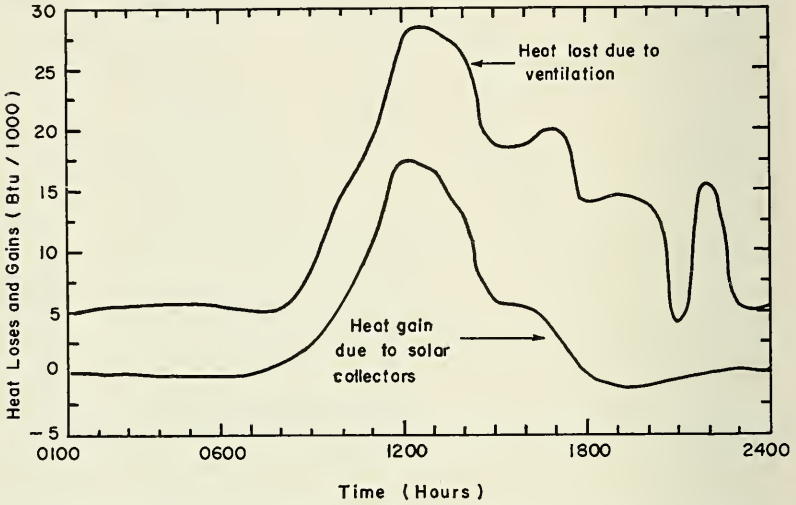

Figure 13. *Heat movement due to ventilation system* (4-4-64).

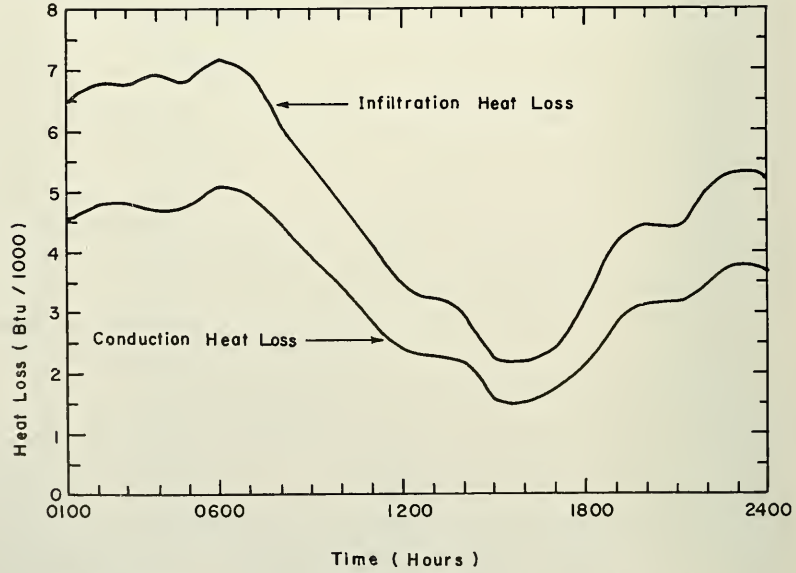

Figure 14. *Heat loss due to infiltration and conduction* (4-4-64).

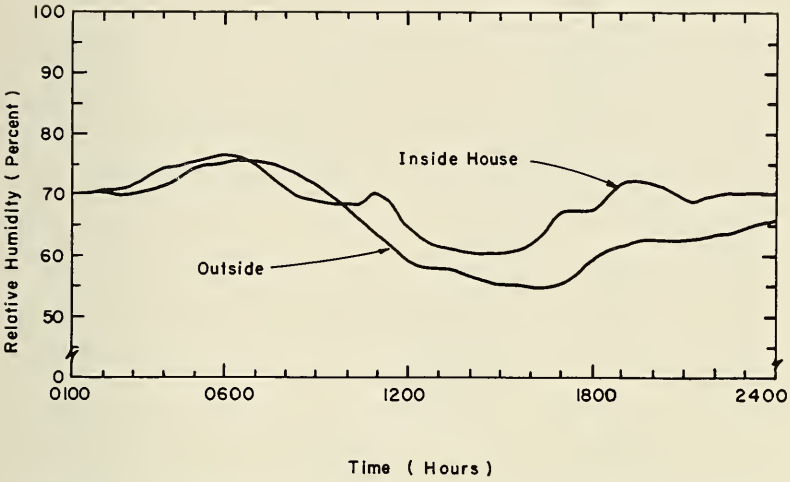

Figure 15. *Comparison of inside and outside relative humidity on a sunny day* (4-4-64).

The second collector had only one plate, but fins were placed 90 degrees to the air flow. This again made a longer path for the air to travel. Finally the third collector was a flat plate collector similar to the one on the poultry house. The characteristics of the collectors are shown in Table 4.

TABLE 4

Solar collector characteristics, October 13, 1965

Description	Collectors	
Type ..... Twin Plate	Finned	Corrugated
Area, sq ft 32.6	32.6	32.6
Cover Sheet ..... Plastic	Plastic	Plastic
Air Passage ..... Over and behind front plate and over second plate	Behind Collector Surface	Behind Collector Surface
Air Flow (cfm) .. 474	225	440

The twin-plated solar collector was more efficient than the other two tested (Figures 17 and 18). On a flat plate collector the maximum rate collected was 180 Btu per square foot. The twin-plate collector absorbed radiation and transferred energy to the air at a maximum rate of 300 Btu per square feet and the finned collector rated between the two.

Figure 18 compares the three collectors on a cloudy day. Even on a cloudy day the double plated collector was more efficient than the flat plate unit.

The double plated collector has not as yet been tested on the poultry house.

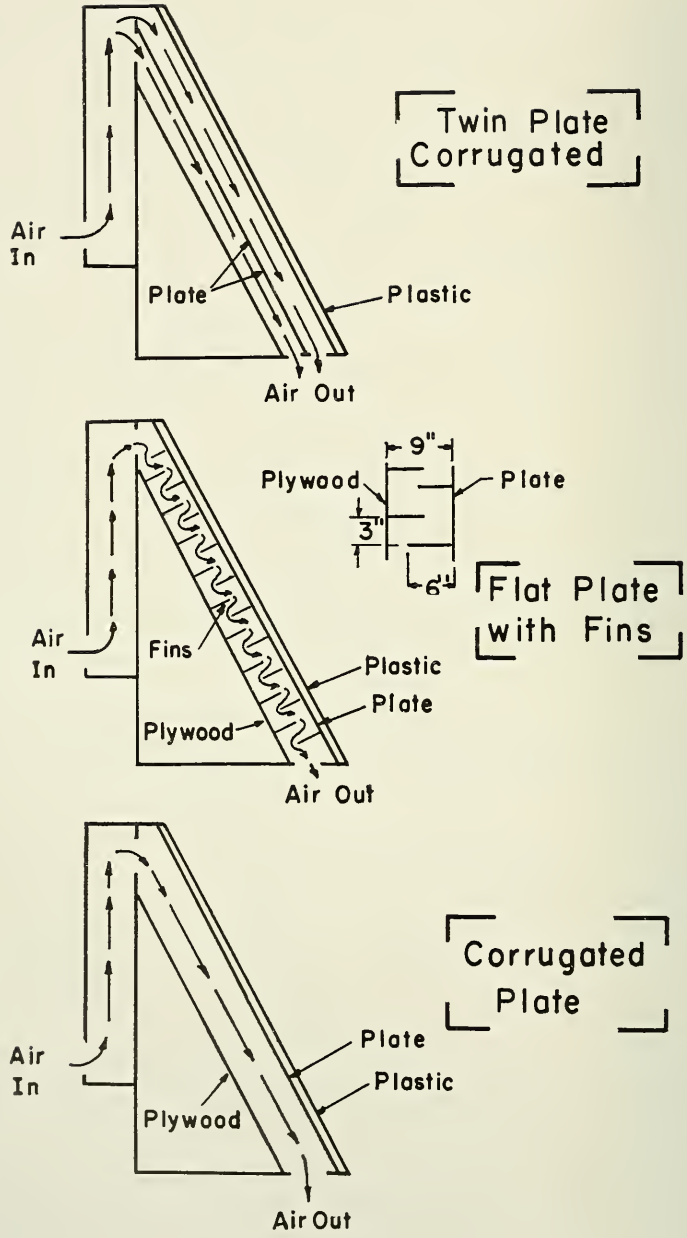

Figure 16. *Cross sections of solar collectors used in* 1965.

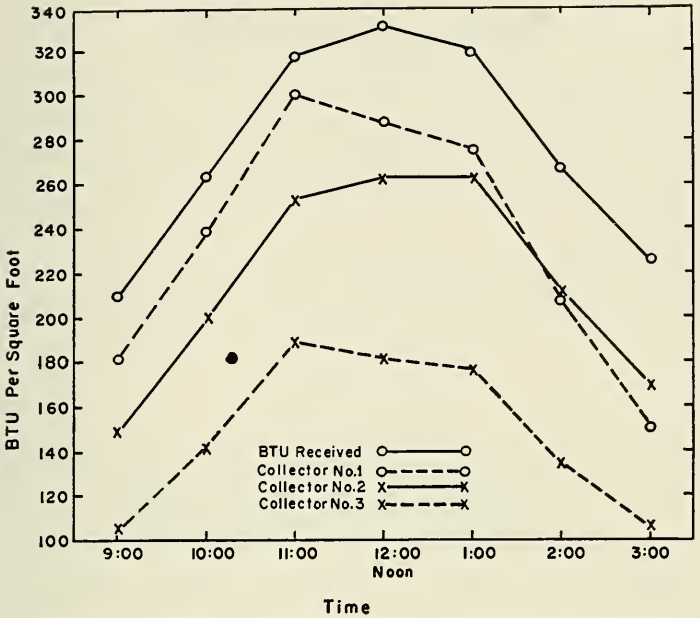

Figure 17. *Energy collected on a sunny day* (3-19-65).

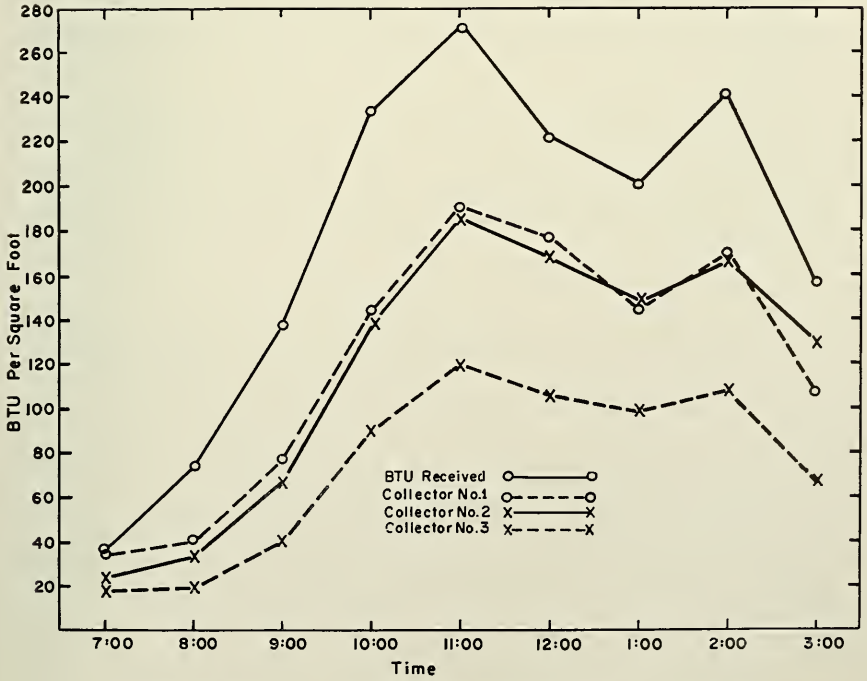

Figure 18. *Energy collected on a cloudy day* (10-14-65).

CONCLUSIONS

Research indicates that the solar collectors are a source of energy which may be used to regulate environment. The collectors decreased the time the outside conditions affected the internal environment of the poultry house. Also, due to the heat addition to the air a higher rate of ventilation was provided which speeded up removal of moisture and ammonia.

Improved collectors were designed which collect at a rate of 300 Btu per square foot, much higher than the original test collectors.

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Appendix

TABLE 1

Calculated heat losses and gains April 4, 1964

	Heat Gains (Btu/hr)				Heat Losses (Btu/hr)		
Time	Hens	Solar Collector	Fans	Lights	Ventilation	Infiltration	Conduction
0100	16,790	0	200	0	5,320	6,540	4,630
0200	16,790	0	200	0	5,493	6,760	4,779
0300	16,790	−90	200	0	5,534	6,760	4,779
0400	16,790	0	200	0	5,712	6,780	4,710
0500	16,790	0	200	0	5,534	6,760	4,780
0600	16,790	0	200	1,800	5,890	7,190	5,100
0700	16,790	400	200	1,800	5,679	6,980	4,934
0800	16,790	700	200	1,800	4,937	6,100	4,320
0900	16,790	2,460	400	1,800	8,780	5,450	3,865
1000	16,790	6,680	740	1,800	14,690	4,800	3,400
1100	16,790	10,120	1,120	1,800	19,230	4,140	2,930
1200	16,790	17,500	1,950	1,800	28,000	3,490	2,480
1300	16,790	17,000	2,110	1,800	28,300	3,270	2,300
1400	16,790	15,000	2,110	1,800	26,300	3,050	2,150
1500	16,790	5,600	2,110	1,800	18,700	2,180	1,540
1600	16,790	5,600	2,110	1,800	18,700	2,180	1,540
1700	1,6790	3,760	2,110	1,800	20,700	2,400	1,710
1800	16,790	0	1,090	1,800	13,600	3,050	2,150
1900	16,790	−1,500	860	1,800	14,800	4,140	2,930
2000	16,790	−1,382	770	0	13,800	4,360	3,080
2100	16,790	−170	200	0	3,470	4,360	3,080
2200	16,790	0	580	0	16,700	5,010	3,550
2300	16,790	−170	200	0	4,180	5,330	3,700
2400	16,790	−170	200	0	4,000	5,230	3,500

TABLE 2

Calculated heat losses and gains April 2, 1964

	Heat Gains (Btu/hr)				Heat Losses (Btu/hr)		
Time	Hens	Solar Collector	Fans	Lights	Ventilation	Infiltration	Conduction
0100	16,790	0	192	0	4,779	5,886	4,160
0200	16,790	0	192	0	4,779	5,886	4,160
0300	16,790	0	192	0	4,956	6,104	4,310
0400	16,790	0	192	0	4,779	5,886	4,160
0500	16,790	0	192	0	4,425	5,450	3,850
0600	16,790	0	192	1,800	4,025	5,014	3,540
0700	16,790	0	416	1,800	8,294	4,796	3,390
0800	16,790	0	448	1,800	9,338	5,014	3,540
0900	16,790	0	704	1,800	10,778	3,706	2,620
1000	16,790	0	768	1,800	8,905	2,834	2,000
1100	16,790	0	1,280	1,800	13,656	2,616	1,950
1200	16,790	0	1,880	1,800	9,954	1,308	925
1300	16,790	0	2,110	1,800	7,256	872	620
1400	16,790	0	2,400	1,800	6,210	654	460
1500	16,790	0	2,830	1,800	12,250	1,090	770
1600	16,790	0	2,110	1,800	9,150	1,090	770
1700	16,790	0	2,110	1,800	7,320	872	620
1800	16,790	0	2,110	1,800	13,020	1,526	1,080
1900	16,790	0	2,110	1,800	9,300	1,090	770
2000	16,790	0	2,110	0	9,300	1,090	770
2100	16,790	0	2,110	0	5,580	654	460
2200	16,790	0	2,110	0	5,529	654	460
2300	16,790	0	2,110	0	3,660	463	300
2400	16,790	0	2,110	0	1,115	218	150

TABLE 3

Poultry house data of April 2, 1964*

Time	Temperature Wet Bulb	Temperature Dry Bulb	Temperature Outside	Temperature under Solar Collector	Relative Humidity Inside	Relative Humidity Outside
0100	46.8	52.1	25.1	23.8	70	75
0200	46.5	51.7	25.3	24.4	70	76
0300	46.4	51.6	23.9	24.7	63	77
0400	46.4	51.5	25.1	24.3	63	76
0500	46.8	52.1	26.5	25.4	70	76
0600	48.2	53.3	29.5	28.7	70	75
0700	47.9	52.7	30.9	31.1	70	74
0800	48.0	52.9	30.3	31.5	70	76
0900	47.1	51.3	34.4	34.8	75	80
1000	47.8	51.5	39.2	29.7	75	81
1100	47.2	51.5	39.9	40.4	73	84
1200	48.7	51.4	45.1	45.5	82	80
1300	52.1	56.3	52.1	53.4	78	76
1400	53.3	58.2	55.1	54.9	75	81
1500	52.9	57.3	52.7	52.3	75	71
1600	52.6	57.0	52.5	51.5	76	71
1700	52.7	56.9	52.5	51.8	76	71
1800	52.9	56.9	50.2	50.7	76	75
1900	53.0	56.6	51.7	50.2	80	76
2000	52.7	56.3	51.2	50 1	79	75
2100	52.0	55.1	51.7	50.4	81	77
2200	52.0	55.5	52.0	51.1	80	75
2300	51.9	55.9	54.3	52.8	79	71
2400	52.2	57.0	56.5	54.8	73	69

*Average of 15 readings per hour.
Temperatures given in degrees F.
Relative humidity given in per cent.

TABLE 4

Poultry house data of April 4, 1964*

Time	Temperature Wet Bulb	Temperature Dry Bulb	Temperature Outside	Temperature under Solar Collector	Relative Humidity Inside	Relative Humidity Outside
0100	47.7	52.5	22.2	22.2	70	70
0200	47.4	52.1	21.3	21.0	70	70
0300	46.4	51.2	20.0	19.5	70	71
0400	46.3	50.8	19.1	19.0	72	74
0500	46.5	50.7	19.0	18.8	74	74
0600	48.2	52.6	19.5	19.6	76	76
0700	49.6	53.9	22.1	24.4	75	75
0800	48.8	53.2	25.0	29.2	74	70
0900	47.7	52.4	27.7	34.9	71	69
1000	46.5	51.9	30.5	40.8	68	68
1100	47.0	52.2	33.5	43.5	70	64
1200	46.0	52.3	35.9	45.5	63	59
1300	47.9	54.6	39.1	48.9	61	58
1400	47.6	54.5	40.3	48.5	60	56
1500	45.9	52.7	42.9	45.8	60	55
1600	44.3	50.3	40.4	43.1	62	55
1700	54.5	51.1	40.6	42.4	67	55
1800	46.5	51.9	37.3	37.3	66	60
1900	47.3	52.0	33.8	31.9	72	62
2000	49.1	53.9	34.0	32.3	71	62
2100	47.0	52.1	32.9	31.1	69	63
2200	48.5	53.7	30.5	29.3	70	65
2300	48.5	53.6	29.9	28.7	70	66
2400	48.5	53.2	29.7	28.5	71	66

*Average of 15 readings per hour.
Temperatures given in degrees F.
Relative humidity given in per cent.